

NASA Contractor Report 181829

Potential Uses of Vacuum Bubbles in Noise and Vibration Control

(NASA-CR-181829) POTENTIAL USES OF VACUUM
BUBBLES IN NOISE AND VIBRATION CONTROL Final
Report (EEN) 53 p CSCI 20A

N89-25674

G3/71 Unclass
0216748

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**Contract NAS1-16521
June 1989**

NASA

National Aeronautics and
Space Administration

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1. INTRODUCTION

BBN Systems and Technologies Corporation (BBN STC) has been tasked by the Acoustics Division of the NASA Langley Research Center (NASA) to investigate the potential of the Vacuum Bubble concept for noise and vibration control with special emphases on spacecraft and aircraft applications. The interest in Vacuum Bubbles is based on their unique property that they are more compliant (i.e., they are softer) than the gas volume they displace.

Scope

The work performed in the framework of this task included: (1) formulation of an analytical model of the Vacuum Bubble in the form of volume compliance, (2) development of a computer program that facilitates the study of how geometric, material and environmental parameters affect the performance, (3) utilization of the computer program to study the effect of the key parameters on performance, (4) development of guidelines for using the computer program for conceptual acoustic design of Vacuum Bubble elements for specific environments, and (5) exploration of the potential application and expected performance of Vacuum Bubbles in noise and vibration control engineering.

Key Findings

The key findings of the study are that:

- (1) Vacuum Bubbles can be designed to have substantially higher compliance than the volume of air they displace.
- (2) As an acoustic element, the Vacuum Bubble is equivalent to a Helmholtz resonator and can be used wherever Helmholtz

resonators proved useful. Their key advantage is that a specified resonance frequency can be achieved with orders of magnitude smaller volume than would be required for traditional Helmholtz resonators.

- (3) The combination of low volume and high compliance makes them potential candidates in low frequency noise control, where traditional Helmholtz resonators were impractical because of the large volume requirements.
- (4) Vacuum Bubbles can be used:
 - for reducing the sound radiation of vibrating surfaces in air or water;
 - for increasing the low frequency sound transmission loss of double walls without increased space requirements;
 - as screens that reflect and absorb an incident sound wave;
 - for achieving substantial attenuation with muffler baffles of small thickness;
 - as a spring of low dynamic but high static stiffness. They require substantially lower volume and static deflection than an equally compliant airspring.
- (5) Groups of differently frequency tuned Vacuum Bubbles can be designed to produce a real acoustic impedance within a limited frequency band. For example, it is possible to design a Vacuum Bubble screen that can match the characteristic impedance of air for plane waves and thereby achieve complete absorption of an incident sound wave in the design frequency band.
- (6) Vacuum Bubbles are made of a thin metallic shell and have vacuum in their cavities and, therefore pose no danger in terms of environmental contamination or fire hazard.

- (7) The acoustical performance of Vacuum Bubbles is very sensitive to variations in static pressure. This is not likely to be a problem in spacecraft applications but may limit their applications in aircraft noise control.
- (8) To assure optimal performance, geometric and material parameters must be controlled to a very high accuracy.
- (9) Based on the above listed findings, it is concluded that Vacuum Bubbles will be more suitable for spacecraft applications than for aircraft applications.

Organization of the Report

Section 2 describes, in a qualitative manner, how Vacuum Bubbles are constructed and how they work. Section 3 documents the analytical model of the Vacuum Bubble and the computer program that has been developed to numerically evaluate the analytical model. The effect of key parameters that influence acoustic performance are discussed in Sec. 4. Section 5 compares Vacuum Bubbles with traditional sound absorbers and points out key advantages and disadvantages. Potential applications and expected performance is discussed in Sec. 6. In Sec. 7 a possible arrangement is discussed that would eliminate the effect of atmospheric pressure variations on acoustic performance.

2. CONCEPT OF VACUUM BUBBLE LINER

The basic construction and the working principle of the Vacuum Bubble are described below. Because Vacuum Bubbles are constructed entirely from metal, they do not pose a fire hazard or contain gas that could accidentally be released. Their key acoustic feature is that they have substantially lower dynamic stiffness than the volume of air they replace. These are all desirable functional and acoustic properties for spacecraft and aircraft noise control applications.

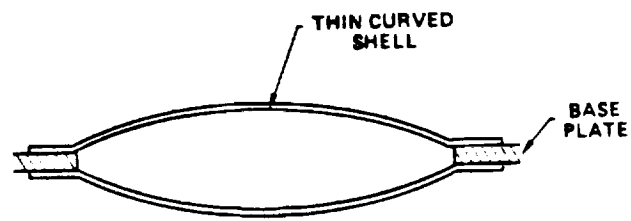
2.1 Construction of the Vacuum Bubble Liner

As shown in Fig. 1, a Vacuum Bubble consists of a square metal plate that has a large round hole punched in its center and of two thin plates which have indentations in the form of a spherical segment. The two thin plates are bonded to the thick plate. When the bonding is done under vacuum, the element is ready for use. When the bonding is done under atmospheric pressure, the cavity must be evacuated before the tile is ready for use. The height of the indentation in the thin shells is highly exaggerated in Figure 1. The actual bubble height is on the order of the magnitude of the shell thickness and thus would not be perceivable if the sketch were drawn to scale.

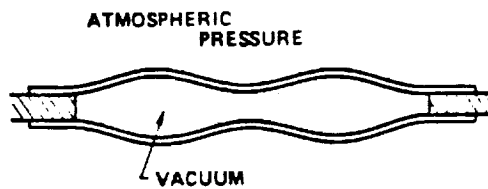
Figure 1(b) shows the shape of the Vacuum Bubble element when it is exposed to atmospheric pressure. Under atmospheric pressure the thin shells deform and the center of the bubble wall deflects, and moves toward the center plate.

2.2 How Does the Vacuum Bubble Liner Work?

This section provides a brief qualitative description of how the Vacuum Bubble liner achieves the high degree of compliance (softness).



(a)



(b)

FIGURE 1. CONCEPTUAL SKETCH OF A VACUUM BUBBLE TILE ELEMENT (CURVATURE OF A SHELL HIGHLY EXAGGERATED).
(a) As Assembled Under Vacuum.
(b) Exposed to Atmospheric Pressure.

Achieving a high compliance with a device of small volume can be done only if (1) the air stiffness within the cavity is eliminated, and (2) the shell of the bubble is dynamically very soft. The first requirement is fulfilled by evacuating the cavity. The second requirement is more difficult to fulfill because the bubble shell must have considerable static stiffness to prevent the bubble from collapsing under the atmospheric pressure. Consequently, the bubble shell must be statically quite stiff, but dynamically very soft. A shell that has the characteristics of such a nonlinear spring fulfills both requirements. This is the key idea behind the proprietary concept of the Vacuum Bubble invented by Dr. Bschorr [1].

The unloaded Vacuum Bubble, as shown in Figure 1(a), is stiff statically as well as dynamically. The stiffness is due to the curvature of the shell. As the static pressure load is applied, the shell deflects and takes up the form shown schematically in Figure 1(b). This deformation pattern of the shell is similar to the deformation pattern of a "Belleville Washer." The deformed shell is near to its buckling point where a small change of load (i.e., the dynamic load represented by the airborne sound) results in a large change in dynamic deflection. In other words, the shell becomes dynamically very soft when the design static pressure load is applied, but remains statically stiff enough not to collapse under the load.

The dynamic load/deflection curve at this optimal working point remains practically linear as long as the dynamic displacement, produced by the incident sound, is less than 10% of the bubble wall thickness [1].

3. DESIGN PARAMETERS, ANALYTICAL MODEL AND COMPUTER PROGRAM

This chapter deals with the key geometric, material and environmental parameters which control the acoustic performance of the Vacuum Bubble. An analytical model for predicting the dynamic stiffness of a Vacuum Bubble and a computer program to perform numerical calculations is presented.

3.1 Design Parameters

The key geometric parameters of the Vacuum Bubble liners, as depicted in Fig. 2(a), are:

s = shell thickness, m

h = bubble height (before evacuation), m

R_0 = bubble radius, m

R = radius of curvature of the indentation, m.

The material parameters are:

ρ = density of the shell material, kg/m³

E = Young's modulus of the shell material, N/m²

μ = Poisson ratio of the shell material.

The environmental design parameter is represented by:

ΔP_{tot} = Pressure differential across the shell.

In our case

$$\Delta P_{tot} = P_{atm}, \text{ N/m}^2$$

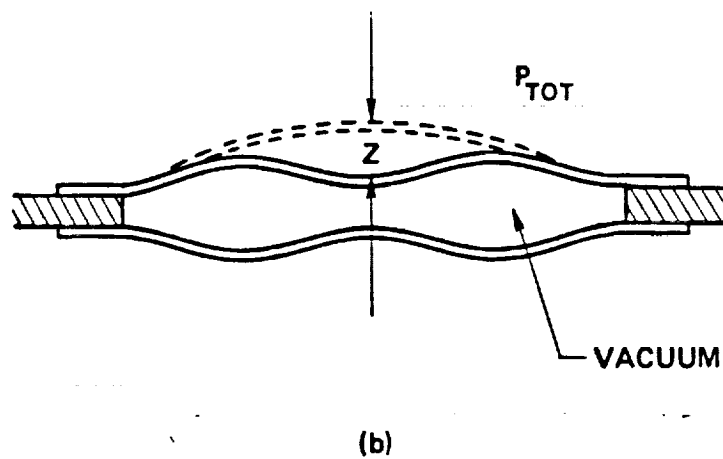
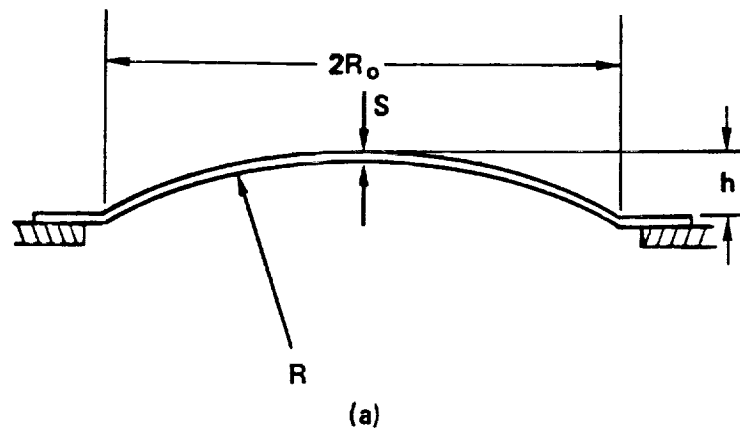


FIGURE 2. GEOMETRIC PARAMETERS OF THE VACUUM BUBBLE TILE UNIT.
 (a) Unloaded Shape (only one side shown).
 (b) Exposed to Atmospheric Pressure.

where P_{atm} is the atmospheric pressure.

The static response parameter is:

z = center displacement of the bubble wall.

This static displacement, which is caused by the total pressure differential across the bubble wall, is shown schematically in Fig. 2(b).

3.2 Analytical Model

The center deflection of the bubble shell, z , depends on applied static pressure load, ΔP_{tot} , as given in Eq. 1 [1].

$$\Delta P_{tot}(z) = E \left(\frac{s}{R_o}\right)^4 \left\{ A \frac{z}{s} + B \left[\left(\frac{z}{s}\right)^3 - 3 \frac{z^2 h}{s^3} + 2 \frac{z h^2}{s^3} \right] \right\}, N/m^2 \quad (1)$$

where

$$A = \frac{16}{3(1-\nu^2)} \quad (2)$$

$$B = \frac{10-6\nu}{3(1-\nu)} - 8/7 \quad (3)$$

and the other symbols are as defined above in Sec. 3.1.

The dynamic stiffness per unit surface area is given by

$$k(z) = \frac{dP_{tot}(z)}{dz} = E \left(\frac{s}{R_o}\right)^4 \left\{ \frac{A}{s} + \frac{B}{s^3} [3z^2 - 6zh + 2h^2] \right\}, N/m^3 \quad (4)$$

Note that $k(z)$ depends on the static center displacement z which in turn depends on the ΔP_{tot} as given in Eq. 1.

The dynamic stiffness has its minimum where $dk(z)/dz = 0$ yielding

$$z = h, \quad (5)$$

According to Eqs. 1 and 5, this occurs where the atmospheric pressure is

$$\Delta P_{\text{tot}} = P_{\text{atm}} = E \left(\frac{s}{R_0} \right)^4 A \frac{h}{s}, \text{ N/m}^2 \quad (6)$$

The geometric and material parameters in Eq. 6 must be chosen specifically so that this pressure coincides with the atmospheric pressure the Vacuum Bubble is designed for. Combining Eqs. 4 and 5 yields

$$k_{\text{min}}(z) = E \left(\frac{s}{R_0} \right)^4 \left[\frac{A}{s} - \frac{B}{s^3} h^2 \right], \text{ N/m}^3. \quad (7)$$

Choice of Wall Thickness and Bubble Height

Equation 7 indicates that zero dynamic stiffness [i.e., $k_{\text{min}}(z) = 0$, and resonance frequency $\omega_0 = 0$] is achievable with the specific choice of geometric and material parameters given in Eq. 8a.

$$h = s \sqrt{\frac{A}{B}} = s \sqrt{\frac{\frac{16}{3(1-\mu^2)}}{\frac{10-6\mu}{3(1-\mu)} - 8/7}}, \text{ m} \quad (8a)$$

For metals like aluminum or steel, where the Poisson ratio is $\mu = 0.3$, Eq. 8a yields as a criteria for zero dynamic stiffness and resonance frequency, $\omega_0 = 0$, at design static pressure

$$h = 1.457s, \text{ m}. \quad (8b)$$

Note that Eqs. 5 and 8b "fix" the relationship between wall thickness s , bubble height h and center displacement z .

Choice of Bubble Radius

After the bubble height, h , and the wall thickness, s , have been selected, the optimum bubble radius, R_{opt} , is obtained by solving Eq. 1 for R_0 with $z/h = 1$, yielding

$$R_{opt} = \left(\frac{EAs^3h}{\Delta P_{tot}} \right)^{1/4}, \text{ m.} \quad (9a)$$

With the optimum choice of the h/s ratio, as given in Eq. 8b, Eq. 9a yields

$$R_{opt} = s \left(\frac{1.457EA}{\Delta P_{tot}} \right)^{1/4}, \text{ m.} \quad (9b)$$

Equivalent Air Layer Thickness

To provide a more familiar measure for the low frequency performance of Vacuum Bubbles than the dynamic stiffness per unit surface area presented in Eq. 4, it is illustrative to express it in the form of the thickness air layer that (for perpendicular sound incidence and for the same surface area, at standard atmospheric pressure, P_0) yields the same volume displacement as the bubble. The equivalent air layer thickness, d_{eq} , is given by

$$d_{eq} = n \frac{P_0 \kappa}{k(z)}, \text{ m} \quad (10)$$

where $k(z)$ is given in Eq. 4, and

$$n = \begin{cases} 0.5 & \text{for single-sided bubbles} \\ 1 & \text{for double-sided bubbles} \end{cases}$$

$$P_0 = \text{standard atmospheric pressure, } 10^5 \text{ N/m}^2$$

$$\kappa = 1.4, \text{ ratio of specific heats of air.}$$

Resonance Frequency of the Vacuum Bubble

The dynamic stiffness, k_B , is given by

$$k_B = R_0^2 \pi k(z), \text{ N/m.} \quad (11)$$

The mass of the moving part of the bubble surface, m_B is

$$m_B = R_0^2 \pi \rho s, \text{ kg.} \quad (12)$$

and the resonance frequency of the Vacuum Bubble resonator, $\omega_0(z)$ is given by

$$\omega_0(z) = \sqrt{k_B/m_B} = \sqrt{k(z)/\rho s}, \quad 1/\text{sec} \quad (13)$$

where ρ is the density of the bubble wall material, s is the wall thickness and $k(z)$ is given in Eq. 4.

Note that the resonance frequency depends on the center static displacement, z , which in turn depends on the static pressure, P_{tot} . Consequently, the resonance frequency also depends on the static pressure. The lowest dynamic stiffness and consequently the lowest resonance frequency is obtained at design atmospheric pressure. Higher or lower than design atmospheric pressures both result in higher dynamic stiffness and higher resonance frequency. This strong dependence of the resonance frequency on static pressure is an undesirable characteristic of the Vacuum Bubble that renders them ineffective for reducing

tonal noise in aircrafts. This is because the change in cabin pressure results in a shift of the resonance frequency.

Acoustic Characteristics of a Single Vacuum Bubble

The acoustic impedance of the Vacuum Bubble, Z_a , is defined as the ratio of the incident sound pressure and the volume velocity response

$$Z_a = p/Q = P/(Su), \text{ Nsec/m}^5. \quad (14)$$

Because the bubble wall motion decreases from center to edge, the volume velocity is given by

$$Q = (1/2) R_o^2 \pi (dz/dt) = j\omega \left(\frac{1}{2}\right) R_o^2 \pi dz, \text{ m}^3/\text{sec} \quad (15)$$

Well below resonance, where $\omega \ll \omega_o$, the bubble response is controlled by the dynamic stiffness of the bubble. At resonance the response is controlled by the resistance of the resonator, R . Above resonance the effective mass of the bubble wall, $m_{eq} = 0.5 R_o^2 \pi \rho$, is the controlling parameter. Accordingly the acoustic impedance takes the following form:

$$Z_a(z) = \frac{1}{S_{eq}} \left(\frac{k(z)}{j\omega} + R + j\omega s\rho \right), \text{ Nsec/m}^5 \quad (16)$$

where $k(z)$ is the dynamic stiffness per unit surface area given in Eq. 4, ρ is the density of the wall material, and $S_{eq} = \frac{1}{2} R_o^2 \pi$. Rearranging Eq. 16 yields

$$Z_a(z) = \frac{s\rho \omega_o(z)}{S_{eq}} \left[\eta - j \left(\frac{\omega_o(z)}{\omega} - \frac{\omega}{\omega_o(z)} \right) \right], \text{ Nsec/m}^5 \quad (17)$$

where $\eta = R/(\rho \omega_0(z) s)$. Equation 17 represents a typical resonance curve. The impedance minimum occurs at the resonance frequency $\omega_0(z)$. Below $\omega_0(z)$ the impedance has stiffness and above it mass character. Note that the resonance frequency, $\omega_0(z)$, depends on the static pressure because z , and $k(z)$ are a function of the static pressure as described in Eqs. 1 and 4. The lowest resonance frequency will occur at that particular atmospheric pressure, ΔP_{tot} , that produces a center displacement $z = h$. This atmospheric pressure can be computed using Eq. 6. Static pressures, both lower and higher than this, will result in higher dynamic stiffness and correspondingly in higher resonance frequency.

The acoustic admittance, $Y_a(z)$, is the inverse of the acoustic impedance given in Eq. 17 and has the form

$$Y_a(z) \equiv \frac{1}{Z_a(z)} = \frac{S_{eq}}{s \rho \omega_0(z)} \frac{\eta + j \left(\frac{\omega_0(z)}{\omega} - \frac{\omega}{\omega_0(z)} \right)}{\eta^2 + \left(\frac{\omega_0(z)}{\omega} - \frac{\omega}{\omega_0(z)} \right)^2}, \frac{m^5}{Nsec} \quad (18)$$

The peak value of the acoustic admittance is $S_{eq}/(s \rho \omega_0(z) \eta)$ and occurs at $\omega = \omega_0(z)$. Measurements of the resonance curve on experimental Vacuum Bubble samples designed for airborne applications [2] yielded a typical value of $\eta = 0.04$. The acoustic impedance and acoustic admittance characterize the acoustic characteristics of the Vacuum Bubble. Note that in addition to the frequency, they also depend on atmospheric pressure.

Acoustic Characteristics of a Vacuum Bubble Group

Most noise sources emit both tonal and broadband noise. As we have shown before, a single Vacuum Bubble can be tuned to provide large compliance at a single frequency. However, the tuning frequency shifts with change in static pressure. Consequently, in all practical cases, a single Vacuum Bubble (or group of them all tuned to the same frequency) is not a viable solution even for a single tone. To deal effectively with broadband noise a group of differently frequency-tuned Vacuum Bubbles is needed. The group must have the following attributes:

- (1) The resonance frequencies of the individual members must be evenly distributed within the frequency range. This is to minimize the number of Vacuum Bubbles needed;
- (2) The spacing between consecutive resonance frequencies, $\Delta\omega_0$, should be smaller than the resonance bandwidth. This is to assure that there is no gap within the frequency range covered;
- (3) The frequency range of Vacuum Bubble tuning at design atmospheric pressure must start at a lower frequency than the lowest frequency of the noise. This is because both positive and negative deviations of the atmospheric pressure from the design static pressure result in an upward shift of the resonance frequency of each individual Vacuum Bubble. This upward shift of frequency coverage with changing static pressure is illustrated in a qualitative manner in Fig. 3.

Groups of differently tuned, independently responding, resonators can have unique properties. One of these unique properties is that the acoustic admittance of a group of resonators can be real and independent of the damping. For

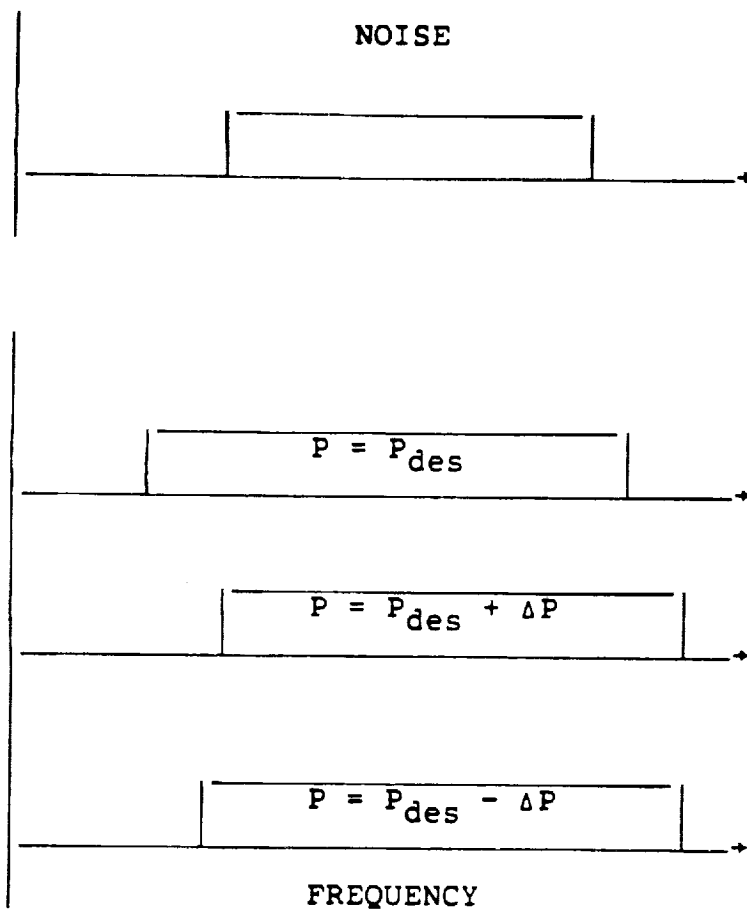


FIGURE 3. CHOICE OF RESONANCE FREQUENCY RANGE TO ASSURE COVERAGE OF THE NOISE BANDWIDTH IN THE STATIC PRESSURE RANGE OF $P_{des} \pm \Delta P$.

Top: Noise Bandwidth
 Bottom: Frequency Range Coverage at Various Static Pressures

example, if the resonance frequencies of resonators within the group is equally spaced (i.e., $\omega_0(n+1) = \omega_0(n) + \Delta\omega_0$ for all the n resonators of the group) and the resonance bandwidth, $n\omega_0$, is larger than $\Delta\omega_0$, then the acoustic admittance of the group of resonators, Y_{ag} is [1]

$$Y_{ag} = \frac{1}{Z_{ag}} = \frac{\pi}{2s\rho S_{eq}} \frac{S_{eq}^2}{\Delta\omega_0} = \frac{S_{eq}}{4s\rho} \frac{1}{\Delta f_0}, \frac{m^5}{Nsec} \quad (19)$$

This is analogous to the result obtained by Cremer and Heckel [2] for the real part of the average mechanical input admittance of a highly damped thin, homogenous, isotropic plate in the frequency region of modal overlap, $Re \{Y_p\}$, which is given by

$$Re \{Y_p\} = \frac{\pi}{2s\rho S_{eq}} n(\omega), \frac{m}{Nsec} \quad (20)$$

where $n(\omega)$ is the modal density of the plate. Considering now that for even frequency spacing $\Delta\omega_0 = 1/n(\omega)$ and the difference in the definition of acoustic and mechanical admittance for resonators that are small compared with the wavelength, namely

$$Y_a = \frac{Q}{P} = \frac{Su}{F/S} = \left(\frac{u}{F}\right) S^2, \frac{m^5}{Nsec} \quad (21a)$$

$$Y_M = \left(\frac{u}{F}\right), \frac{m}{Nsec} \quad (21b)$$

We note that Eqs. 19 and 20 are formally equivalent. The physical reason for this equivalency is that in the case of a thin plate in bending each resonant mode can be considered as a separate single degree-of-freedom mechanical resonator characterized by its modal mass and modal stiffness. The reason

that the frequency average admittance does not depend on the damping is, that for lightly damped resonators the power at resonance is inversely proportional to the damping but the resonance bandwidth is directly proportional to the damping. Consequently, the power dissipated in the resonator when the excitation is a broadband random sound pressure or force is practically independent of the damping.

Equation 19 can be used to approximate the number of differently tuned Vacuum Bubbles that are needed to yield any desired real acoustic admittance in a specified frequency band of width Δf . Considering that the number of differently tuned Vacuum Bubbles needed to cover the frequency range Δf is $\Delta f / \Delta f_0$ and that the total active surface of the group is $S_g = S_{eq} (\Delta f / \Delta f_0)$, Eq. 19 can be written in the form

$$Y_{ag} = \frac{S_g}{4s\rho \Delta f} , \frac{m^5}{Nsec} , \quad (22)$$

indicating that to achieve a large compliance requires a small $s\rho \Delta f$ product. Consequently, the mass per unit area of the bubble wall shall be as small as possible to achieve a high compliance in a wide frequency range.

As an example for using Eq. 22 consider the following problem: If the Vacuum bubbles are made of 0.125mm (1/200 inch) thick aluminum, and the Vacuum Bubble screen has 50% active surface area, what is the bandwidth for which the screen can produce a real admittance that equals the plane wave acoustic admittance of the air at standard conditions, namely, $Y_{ag} = Y_{air} = \frac{1}{\rho_0 c_0}$ per meter square.

$$= \rho_{alv} s = 2700 \text{ kg/m}^3 \times 1.25 \times 10^{-4} \text{ m} = 0.34 \text{ kg/m}^2$$

$$S_g = 0.5 \text{ m}^2$$

$$Y_{\text{air}} = \frac{\text{m}^2}{340 \text{ m/sec} \times 1.2 \text{ kg/m}^3} = \frac{1}{408} \frac{\text{m}^4 \text{ sec}}{\text{kg}}$$

Solving Eq. 22 for Δf with the above values yields

$$\Delta f = \frac{S_g}{4\rho_s Y_{\text{ag}}} = \frac{0.5 \text{ m}^2}{4 \times 0.34 \frac{\text{kg}}{\text{m}^3} \times \frac{1}{408} \frac{\text{m}^4 \text{ sec}}{\text{kg}}} = 150 \text{ Hz}$$

Analytical Formulation for Design Purposes

Equation 1 does not lend itself for evaluating the effect of change in the atmospheric pressure (i.e., the outside pressure in our case) on the dynamics of the Vacuum Bubble because it yields the pressure differential, ΔP_{tot} as a function of the bubble center displacement z , in the form $\Delta P_{\text{tot}}(z)$. For design purposes we need the inverse relationship; namely the function $z(\Delta P_{\text{tot}})$. Because Eq. 1 is a third order equation in z we must solve this third order equation.

After some rearrangement, Eq. 1 can be written in the following form:

$$z'^3 - pz'^2 + qz' + r = 0 \tag{23a}$$

where

$$z' = z/h \tag{23b}$$

$$a = hs \quad (23c)$$

$$p = -3 \quad (23d)$$

$$q = (2 + \frac{A/B}{a^2}) \quad (23e)$$

$$r = - \Delta P_{tot} \frac{aR_o^4}{h^4 BE} \quad (23f)$$

Eq. 23a can be further simplified to take the form

$$x^3 + 2x - b = 0 \quad (24a)$$

by substituting $z' = (x - p/3)$ into Eq. 23a.

where

$$a = \frac{1}{3} (3q - p^2) = (\frac{A/B}{a^2} - 1) \quad (24b)$$

$$b = \frac{1}{27} (2p^3 - 9pq + 27r) = \frac{A/B}{a^2} + r \quad (24c)$$

The solution takes the form

$$x = Y + Z \quad (25a)$$

where

$$Y = \{-\frac{b}{2} + \frac{b}{2} (1 + \frac{4}{27} \frac{a^3}{b^2})^{1/2}\}^{1/3} \quad (25b)$$

$$Z = \{-\frac{b}{2} - (\frac{b^2}{4} + \frac{a^3}{27})^{1/2}\}^{1/3}, \quad (25c)$$

and z is finally obtained as $z = z'/h = (x - p/3)/h$. The LOTUS 123 program, named "NASA3BUBB", presented in the next section, incorporates the above described tedious solution of the third order equation (Eq. 1).

3.3 "NASA3BUBB" Computer Program

"NASA3BUBB" occupies the region A-1 to S-61 on the LOTUS 123 Spreadsheet presented on pages 22 and 23. The region A-1 to I-61 is used for individual computations, while the region J-1 to S-61 is to calculate the compliance of up to ten differently pressure-tuned double-sided bubble sheets utilizing the MACRO "/a".

Input Variables

The input variables are entered into the range B3 to B14. The program is currently set up for Aluminum that has the following material properties:

Density, $m = 2700 \text{ kg/m}^2$	[B3]
Elasticity Mod., $E = 7.16 \times 10^{10} \text{ N/m}^2$	[B5]
Poisson Ratio, $m = 0.3$	[B6]

The primary input parameters to be entered are:

Design Stat. Pressure dP , in N/m^2	[B4]
Bubble Wall Thickness, s , in mm	[B7]
Bubble Height, h , in mm (to be chosen $1.44 < h/s < 1.457$)	[B8]
Bubble Radius, R_b , in mm (to be entered from field E13 utilizing the /RV E13 B13 command, to assure that pressure-tuning occurs at design static pressure)	[B13]

Output

The output of the program occupies the region A21 to I61.

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INSTRUMENT COMPUTERIZED DESIGN PROGRAM 6/30/86									
99000									
INPUT PARAMETERS									
Mat. Resistivity, ρ	2700	kgm ³	2700 kgm ³ for Aluminum						
Press. Diff., ΔP	101250	N/m ²	1 atm=1.01325 N/m ²						
Youngs Modulus, E	7.16E+10	N/m ²							
Poisson Ratio, ν	0.30								
Wall Thickness, s	0.250	mm							
Bubble Height, h	0.36	mm							
RLP	1.44								
R	5.061								
B	2.762								
C	4.80E-07								
Bubble Radius, Rb	12.357	mm							
Height, Parn., t	0.585								
Macro V Above Calc for the Tuning Pres									
Normalized Center Displacement, Zn=2/h (Dimensionless)									
Press. Fo									
Dynamic Equ. for Stiffness, K(Zn)									
Approximate Resonance Frequency, Hz									
12.426									
0.756	99350.0	0.020214	0.010130	99350.0	5.55E-07	2.6	0.10	1726	11.8
0.759	99300.0	0.018708	0.009678	99300.0	5.45E-07	2.6	0.10	1710	10.4
0.761	99350.0	0.019205	0.009628	99350.0	5.35E-07	2.7	0.10	1694	9.3
0.764	99400.0	0.018696	0.009574	99400.0	5.24E-07	2.7	0.11	1677	8.5
0.766	99450.0	0.018192	0.009522	99450.0	5.14E-07	2.8	0.11	1661	7.6
0.769	99500.0	0.017687	0.009470	99500.0	5.03E-07	2.8	0.11	1644	7.2
0.772	99550.0	0.017182	0.009418	99550.0	4.93E-07	2.9	0.11	1626	6.7
0.775	99600.0	0.016678	0.009366	99600.0	4.82E-07	2.9	0.12	1609	6.3
0.778	99650.0	0.016171	0.009314	99650.0	4.72E-07	3.0	0.12	1591	5.9
0.781	99700.0	0.015666	0.009263	99700.0	4.61E-07	3.1	0.12	1572	5.6
0.784	99750.0	0.015160	0.009211	99750.0	4.50E-07	3.2	0.12	1553	5.3
0.787	99800.0	0.014655	0.009159	99800.0	4.39E-07	3.2	0.13	1534	5.1
0.790	99850.0	0.014150	0.009108	99850.0	4.27E-07	3.3	0.13	1515	4.8
0.793	99900.0	0.013644	0.009056	99900.0	4.16E-07	3.4	0.13	1494	4.6
0.797	99950.0	0.013139	0.009005	99950.0	4.05E-07	3.5	0.14	1474	4.5
0.800	100000.0	0.012633	0.008954	100000.0	3.93E-07	3.6	0.14	1452	4.3
0.804	100050.0	0.012128	0.008903	100050.0	3.81E-07	3.7	0.15	1431	4.1
0.808	100100.0	0.011623	0.008852	100100.0	3.70E-07	3.8	0.15	1408	4.0
0.811	100150.0	0.011117	0.008801	100150.0	3.58E-07	4.0	0.16	1385	3.9
0.815	100200.0	0.010612	0.008750	100200.0	3.45E-07	4.1	0.16	1361	3.7
0.819	100250.0	0.010107	0.008699	100250.0	3.33E-07	4.3	0.17	1337	3.6
0.824	100300.0	0.009601	0.008648	100300.0	3.21E-07	4.4	0.17	1312	3.5
0.828	100350.0	0.009096	0.008597	100350.0	3.08E-07	4.6	0.18	1285	3.4
0.833	100400.0	0.008591	0.008546	100400.0	2.95E-07	4.8	0.19	1258	3.3
0.837	100450.0	0.008085	0.008495	100450.0	2.82E-07	5.0	0.20	1230	3.3
0.843	100500.0	0.007580	0.008444	100500.0	2.69E-07	5.3	0.21	1201	3.2
0.848	100550.0	0.007075	0.008393	100550.0	2.55E-07	5.6	0.22	1170	3.1
0.853	100600.0	0.006569	0.008342	100600.0	2.41E-07	5.9	0.23	1136	3.0
0.859	100650.0	0.006064	0.008291	100650.0	2.27E-07	6.2	0.25	1104	3.0
0.866	100700.0	0.005558	0.008240	100700.0	2.13E-07	6.7	0.26	1069	2.9
0.872	100750.0	0.005053	0.008189	100750.0	1.98E-07	7.2	0.28	1032	2.8
0.880	100800.0	0.004548	0.008138	100800.0	1.83E-07	7.7	0.30	992	2.8
0.888	100850.0	0.004042	0.008087	100850.0	1.68E-07	8.4	0.33	950	2.7
0.896	100900.0	0.003537	0.008036	100900.0	1.53E-07	9.3	0.37	906	2.7
0.906	100950.0	0.003032	0.007984	100950.0	1.37E-07	10.3	0.41	856	2.6
0.917	101000.0	0.002526	0.007933	101000.0	1.22E-07	11.7	0.46	807	2.6
0.929	101050.0	0.002021	0.007881	101050.0	1.06E-07	13.4	0.53	751	2.5
0.943	101100.0	0.001516	0.007830	101100.0	9.10E-08	15.6	0.61	695	2.5
0.960	101150.0	0.001010	0.007779	101150.0	7.77E-08	18.3	0.72	646	2.4
0.975	101200.0	0.000505	0.007728	101200.0	6.79E-08	20.5	0.82	604	2.4
1.000	101250.0	5.0E-16	0.007677	101250.0	6.42E-08	22.1	0.87	587	2.4

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K	L	M	N	O	P	Q	R	S
99250	99500	99750	100000	100250	100500	100750	101000	101250
List Desired Tuning Pressures in the Range J1 to S1 Above								
RV"013"(6010)013"/C"J10"/RV621.661"J21"								
RV"013"(6010)013"/C"L10"/RV621.661"K21"								
RV"013"(6010)013"/C"J10"/RV621.661"J21"								
RV"013"(6010)013"/C"K10"/RV621.661"K21"								
RV"013"(6010)013"/C"L10"/RV621.661"K21"								
RV"013"(6010)013"/C"O10"/RV621.661"O21"								
RV"013"(6010)013"/C"P10"/RV621.661"P21"								
RV"013"(6010)013"/C"Q10"/RV621.661"Q21"								
RV"013"(6010)013"/C"R10"/RV621.661"R21"								
RV"013"(6010)013"/C"S10"/RV621.661"S21"								
states the Equivalent Air Layer Thickness								
values Entered in the Ranges J1 to S1								

Air Layer Thickness in mm								
For Rb in mm								
12.41E	12.411	12.403	12.395	12.387	12.380	12.372	12.364	12.357

22.6	11.7	7.2	5.3	4.3	3.6	3.2	2.8	2.6
21.3	13.5	7.6	5.6	4.4	3.7	3.2	2.9	2.6
18.5	15.7	8.5	5.9	4.6	3.8	3.3	2.9	2.7
15.6	18.5	9.3	6.3	4.8	4.0	3.4	3.0	2.7
13.5	21.2	10.4	6.7	5.0	4.1	3.5	3.1	2.8
11.8	22.5	11.7	7.2	5.3	4.3	3.6	3.2	2.8
10.4	21.2	13.5	7.8	5.6	4.4	3.7	3.2	2.9
9.3	18.5	15.7	8.5	5.9	4.6	3.8	3.3	2.9
8.5	15.7	18.5	9.3	6.3	4.8	4.0	3.4	3.0
7.8	13.5	21.2	10.4	6.7	5.0	4.1	3.5	3.1
7.2	11.7	22.4	11.7	7.2	5.3	4.3	3.6	3.2
6.7	10.4	21.2	13.5	7.8	5.6	4.4	3.7	3.2
6.3	9.3	18.5	15.7	8.5	5.9	4.6	3.8	3.3
5.9	8.5	15.7	18.4	9.3	6.3	4.8	4.0	3.4
5.6	7.8	13.5	21.1	10.4	6.7	5.0	4.1	3.5
5.3	7.2	11.7	22.4	11.7	7.2	5.3	4.3	3.6
5.1	6.7	10.4	21.1	13.4	7.7	5.6	4.4	3.7
4.6	6.3	9.3	18.4	15.7	8.4	5.9	4.6	3.8
4.6	5.9	8.5	15.7	18.4	9.3	6.3	4.6	4.0
4.4	5.6	7.8	13.5	21.1	10.4	6.7	5.0	4.1
4.3	5.3	7.2	11.7	22.3	11.7	7.2	5.3	4.3
4.1	5.1	6.7	10.4	21.1	13.4	7.7	5.6	4.4
4.0	4.8	6.3	9.3	18.4	15.7	8.4	5.9	4.6
3.9	4.6	5.9	8.5	15.7	18.4	9.3	6.2	4.8
3.7	4.4	5.6	7.8	13.4	21.0	18.4	6.7	5.0
3.6	4.3	5.3	7.2	11.7	22.3	11.7	7.2	5.3
3.5	4.1	5.1	6.7	10.4	21.0	13.4	7.7	5.6
3.4	4.0	4.8	6.3	9.3	18.4	15.6	8.4	5.9
3.3	3.9	4.6	5.9	8.5	15.7	18.3	9.3	6.2
3.3	3.7	4.4	5.6	7.8	13.4	21.0	18.3	6.7
3.2	3.6	4.3	5.3	7.2	11.7	22.2	11.7	7.2
3.1	3.5	4.1	5.0	6.7	18.4	21.0	13.4	7.7
3.0	3.4	4.0	4.8	6.3	9.3	18.3	15.6	8.4
3.0	3.3	3.9	4.6	5.9	8.4	15.6	18.3	9.3
2.9	3.2	3.7	4.4	5.6	7.7	13.4	28.9	18.3
2.8	3.2	3.6	4.3	5.3	7.2	11.7	22.2	11.7
2.8	3.1	3.5	4.1	5.0	6.7	18.4	28.9	13.4
2.7	3.0	3.4	4.0	4.8	6.3	9.3	18.3	15.6
2.7	3.0	3.3	3.9	4.6	5.9	8.4	15.6	18.3
2.6	2.9	3.2	3.7	4.4	5.6	7.7	13.4	28.9
2.6	2.8	3.2	3.6	4.3	5.3	7.2	11.7	22.1

Column B21, B61

Column B21, B61 contains the present range of atmospheric pressure to be used. It starts at 99250 N/m^2 and progresses in 50 N/m^2 steps to 101250 N/m^2 . This is the range where the atmospheric pressure is 95% of the time. If needed, the range can be extended by entering new numbers using the Data Fill command.

Column A21, A61

Column A21, A61 lists the normalized bubble displacement, z/h , obtained by solving the cubic equation (Eq. 19a) describing the static pressure vs. bubble center displacement relationship for the particular values of atmospheric pressure listed in Column B21, B61.

Columns C21, C61 and D21, D61

Columns C21, C61 and D21, D61 contain terms needed to solve the cubic equation.

Column E21, E61

Column E21, E61 lists the atmospheric pressure that is needed to obtain the normalized center displacement values listed in Column A21, A61. If the cubic equation (Eq. 23a) is solved properly, Columns B21, B61 and E21, E61 should be identical. Actually, Column E21, E61 has the sole purpose of checking whether the cubic equation has been solved properly.

Column F21, F61

Column F21, F61 lists the dynamic stiffness per unit surface area of a double-sided* bubble sheet at the atmospheric pressure listed in Column B21, B61.

Columns G21, G61 and H21, H61

Columns G21, G61 and H21, H61 list the equivalent air layer thickness as defined in Eq. 10 in millimeter and in inch, respectively.

Range J1 to S1

Range J1 to S1 is designated for entering tuning atmospheric pressures for which MACRO "/a", starting at H3, automatically computes the required bubble radius, R_b , enters it into the range B13 and J18 to S18 and for these specific values of R_b calculates the equivalent air layer thicknesses and enters them into the Columns J21 and S21.

Experience indicates that $ALP = h/s = 1.44$ and a tuning pressure stepping of 250 N/m^2 results in a smooth compliance vs. atmospheric pressure characteristics.

Example

The printout on page 26 illustrates the use of the program. First, let's illustrate how the program works for computing the properties of a Vacuum Bubble to be pressure-tuned at 10^5 N/m^2 . The pressure tuning curve is plotted in Fig. 4 on page 27. The steps to be undertaken are listed below:

- Enter 10^5 into B4
- Enter 0.25 into B7 ($s=0.25 \text{ mm}$ is arbitrary choice)
- Enter $h=1.44*0.25$ into B8 ($h/s=1.44$ yields desirable tuning curve; see Fig. 4)

*For one-sided bubble sheet these values must be doubled.

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"MRSABUB" COMPUTERIZED DESIGN PROGRAM 6/6/86									
INPUT PARAMETERS									
Mat. Density, ρ	2700	kg/m ³	2700	kg/m ³	for Aluminum				
Press. Diff., ΔP	100000	N/mq.m	1	Atm=1.01321E+5	N/mq.m=14.7 psi				
Youngs Modulus, E	7.14E+10	N/mq.m							
Poisson Ratio, μ	0.30	-							
Wall Thickness, s	0.250	mm							
Bubble Height, h	0.36	mm							
R/P	1.44	-							
R	5.661								
B	2.762								
E	4.80E+07								
Bubble Radius, Rb	12.357	mm							
Height, Farn., 1	0.985	-							
Normalized Center Displacement, Z_{max}/h									
(Dimensionless)									
Press. Po									
Preset									
N/mq.m									
b									
(b ² /4)									
Press. Po									
Calculated									
N/mq.m									
Dynamic Equ. Air									
Stiffness/Layer									
K(Zn)									
Thick.									
mm									
0.756	99250.0	0.020214	0.010130	99250.0	5.55E+07				2.6
0.755	99300.0	0.019708	0.009678	99300.0	5.45E+07				2.6
0.761	99350.0	0.019203	0.009262	99350.0	5.35E+07				2.7
0.764	99400.0	0.018696	0.0088374	99400.0	5.24E+07				2.7
0.766	99450.0	0.018182	0.008422	99450.0	5.14E+07				2.8
0.765	99500.0	0.017667	0.008007	99500.0	5.03E+07				2.8
0.772	99550.0	0.017162	0.007592	99550.0	4.93E+07				2.9
0.775	99600.0	0.016656	0.007176	99600.0	4.82E+07				2.9
0.776	99650.0	0.016151	0.006761	99650.0	4.72E+07				3.0
0.781	99700.0	0.015646	0.006346	99700.0	4.61E+07				3.1
0.784	99750.0	0.015140	0.005931	99750.0	4.50E+07				3.2
0.787	99800.0	0.014635	0.005516	99800.0	4.39E+07				3.2
0.790	99850.0	0.014130	0.005100	99850.0	4.27E+07				3.3
0.793	99900.0	0.013624	0.004685	99900.0	4.16E+07				3.4
0.797	99950.0	0.013119	0.004269	99950.0	4.05E+07				3.5
0.800	100000.0	0.012613	0.003854	100000.0	3.93E+07				3.6
0.804	100050.0	0.012108	0.003438	100050.0	3.81E+07				3.7
0.806	100100.0	0.011602	0.003023	100100.0	3.70E+07				3.8
0.811	100150.0	0.011097	0.002607	100150.0	3.58E+07				4.0
0.815	100200.0	0.010591	0.002192	100200.0	3.46E+07				4.1
0.819	100250.0	0.010086	0.001776	100250.0	3.33E+07				4.3
0.824	100300.0	0.009580	0.001361	100300.0	3.21E+07				4.4
0.826	100350.0	0.009074	0.000945	100350.0	3.08E+07				4.6
0.833	100400.0	0.008568	0.000530	100400.0	2.95E+07				4.8
0.837	100450.0	0.008062	0.000114	100450.0	2.82E+07				5.0
0.843	100500.0	0.007556	0.000000	100500.0	2.69E+07				5.3
0.846	100550.0	0.007050	0.000000	100550.0	2.55E+07				5.6
0.853	100600.0	0.006544	0.000000	100600.0	2.41E+07				5.9
0.855	100650.0	0.006038	0.000000	100650.0	2.27E+07				6.2
0.866	100700.0	0.005532	0.000000	100700.0	2.13E+07				6.7
0.872	100750.0	0.005026	0.000000	100750.0	1.98E+07				7.2
0.880	100800.0	0.004520	0.000000	100800.0	1.83E+07				7.7
0.888	100850.0	0.004014	0.000000	100850.0	1.68E+07				8.4
0.896	100900.0	0.003508	0.000000	100900.0	1.53E+07				9.3
0.906	100950.0	0.003002	0.000000	100950.0	1.37E+07				10.3
0.917	101000.0	0.002526	0.000000	101000.0	1.22E+07				11.7
0.929	101050.0	0.002020	0.000000	101050.0	1.06E+07				13.4
0.943	101100.0	0.001514	0.000000	101100.0	9.10E+06				15.6
0.960	101150.0	0.001010	0.000000	101150.0	7.77E+06				18.3
0.975	101200.0	0.000505	0.000000	101200.0	6.79E+06				28.9

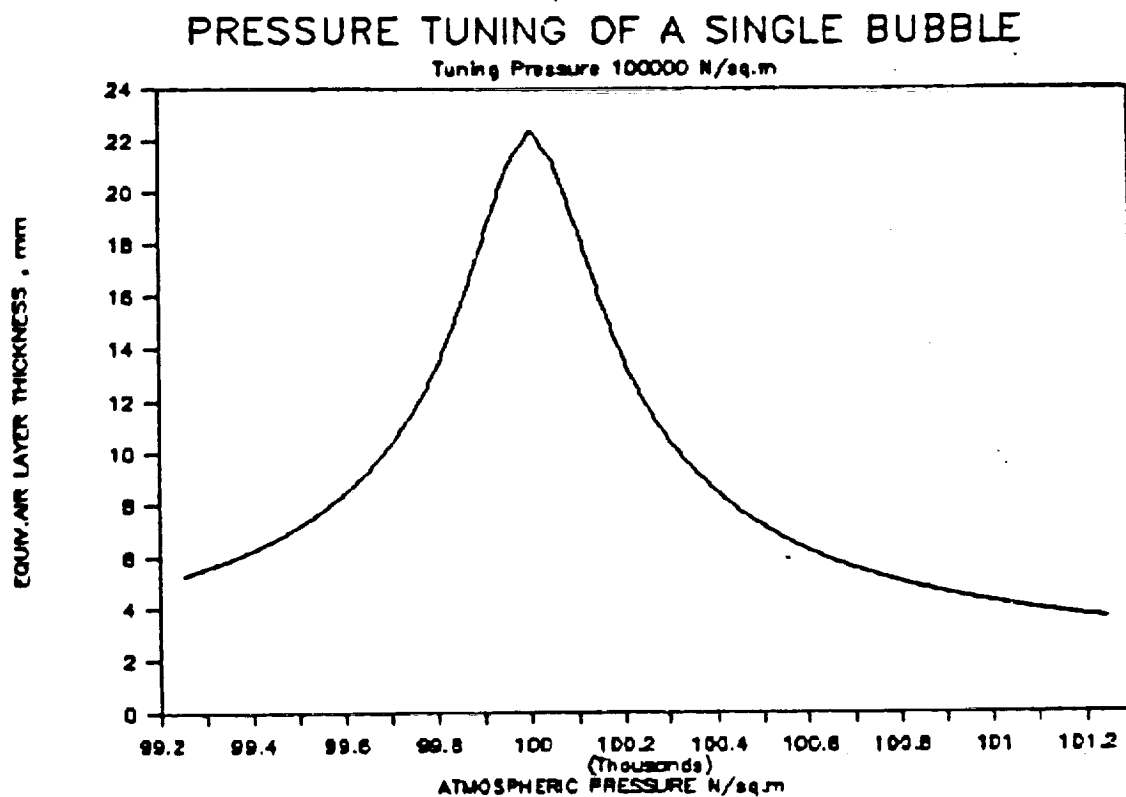


FIGURE 4: PLOT OF THE PRESSURE TUNING CHARACTERISTICS OF THE DOUBLE-SIDED BUBBLE SHEET COMPUTED USING THE WORKSHEET PRESENTED ON PAGE 26.

- Needed Rb (12.395) is displayed automatically in E13
- Enter this value into B13
- At this time the program calculates the worksheet in the Range A1, H61. The completed worksheet is that presented on page 26.

MACRO "/a" performs the same calculations automatically for up to ten preset values of tuning atmospheric pressure entered in the space J1 to S1 at the top of the worksheet. In this example, the preset pressure range is from 99000 N/m² to 101250 N/m², in 250 N/m² increments. The required bubble radius to achieve tuning at the preset atmospheric pressures are entered by the computer into the appropriate ranges from J18 to S18. The MACRO, as presently set up, lists only the Equivalent Air Layer Thickness in Columns J21-J61 to S21-S61 as illustrated in the printouts presented in pages 22 and 23. The individual compliance vs. atmospheric pressure data, listed in these columns, can then be easily plotted or averaged as desired. Figure 5 shows the data in graphical form. The lower graph depicts each of the six* individual tuning curves, while the upper graph shows the average of the six individual tuning curves. Figure 6 has been computed for ten differently tuned bubble sheets. Note that the judicious choice of the h/s ratio and tuning pressure spacing yielded a smooth average performance curve.

*Because the plotting routine of LOTUS 123 can handle only 6 curves.

AVERAGE COMPLIANCE OF SIX BUBBLE SHEETS

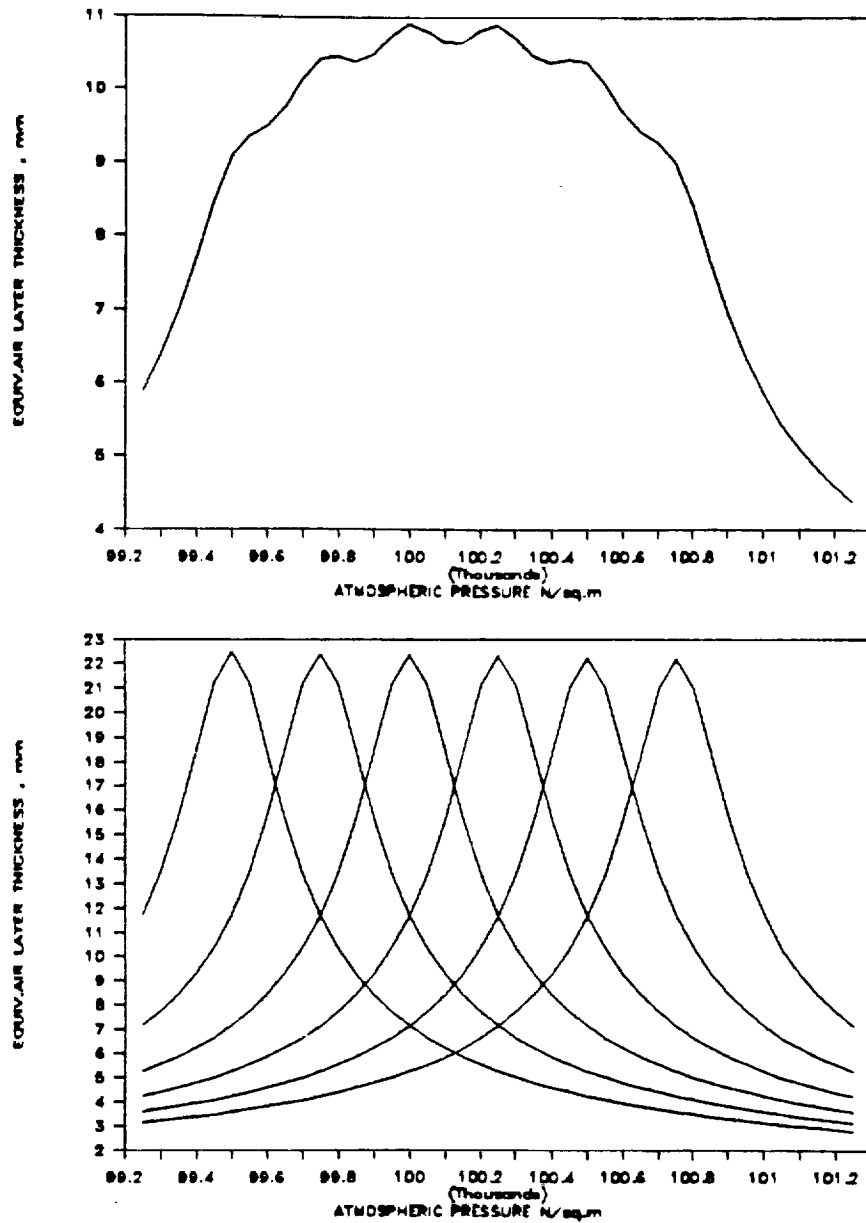


FIGURE 5: PRESSURE TUNING CHARACTERISTICS OF SIX DIFFERENTLY TUNED BUBBLE SHEETS;

Top: Average Compliance Per Unit Surface Area

Bottom: Individual Compliance of Each Sheet

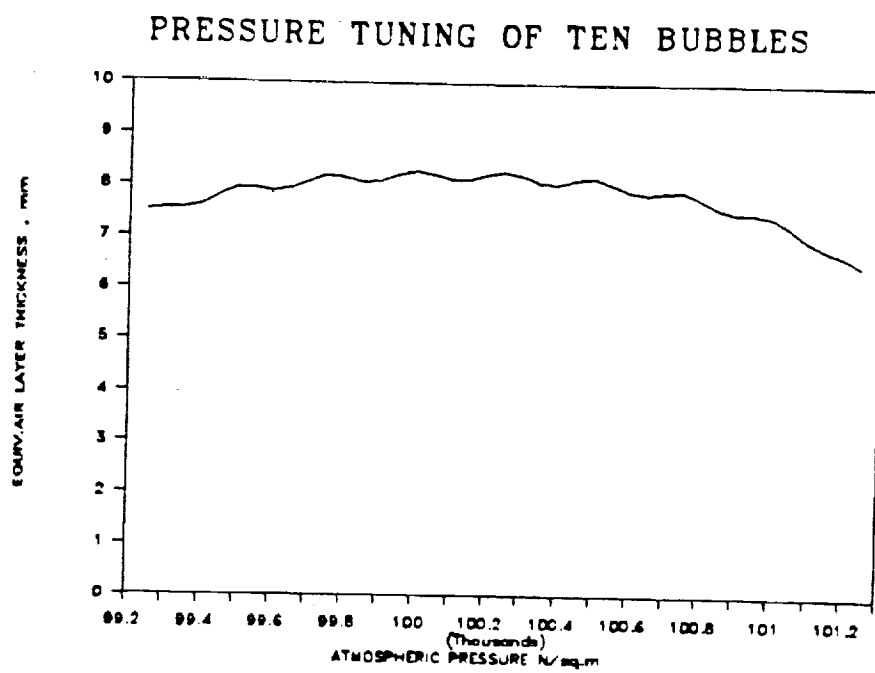


FIGURE 6: PRESSURE TUNING OF TEN BUBBLES.

4. KEY PARAMETERS INFLUENCING PERFORMANCE

A computer analysis has been carried out to investigate how environmental and geometric parameters influence performance. The performance is measured in the form of the equivalent air layer thickness that has the same stiffness as the stiffness component of the impedance of the Vacuum Bubble. It is plotted as a function of atmospheric pressure. Though this characterization is strictly valid only well below the resonance frequency of the Vacuum Bubble, it was chosen to facilitate an overview. Naturally, the mass component of the bubble impedance also must be accounted for. Since the mass component does not depend on static pressure, it has been dropped from further consideration.

4.1 Effect of Atmospheric Pressure

Figure 4 on page 27 shows a typical compliance vs. atmospheric pressure curve computed for a Vacuum Bubble designed for 10^5 N/m^2 atmospheric pressure. As expected, the compliance peaks at this atmospheric pressure and decreases rapidly when the static pressure becomes smaller or larger than this design value. The static pressure change from $0.99 \times 10^5 \text{ N/m}^2$ to 1.01 N/m^2 on the horizontal scale corresponds to a range encountered at sea level due to weather changes. Even this weather-induced changes of atmospheric pressure are large enough to render a single Vacuum Bubble ineffective. Consequently, a group of Vacuum Bubbles which are each "tuned" (or designed for) slightly different atmospheric pressure is the only feasible way to assure effectiveness over a reasonable atmospheric pressure range. The performance of such differently pressure tuned Vacuum Bubble groups have been computed and are plotted in Figs. 5 and 6. In practical applications the manufacturing inaccuracies are likely to result in random pressure tuning of nominally identical Vacuum Bubbles.

4.2 Effect of the h/s Ratio on the Shape of the Tuning Curve

Figure 7 shows the effect of changing the Bubble Height, h , on the shape of the pressure tuning curve. The bubble radius, R_b , has been adjusted to retain the pressure tuning at 100000 N/m^2 .

Observing Fig. 7, note that the peak of the compliance vs. atmospheric pressure curves increases dramatically as the $ALP = h/s$ ratio approaches the critical value of 1.457, where (theoretically) the peak compliance would be infinite. However, outside of the peak region (i.e., at 2% higher or lower than the tuning pressure) the effect of the h/s Ratio is less pronounced.

Because thermal expansion and contraction of the bubble wall results in change of Bubble Height, it is expected to have an effect similar to the change of bubble height.

4.3 Effect of Change in Bubble Radius

Figure 8 shows the effect the change in the Bubble Radius, R_b , has on performance. Each of the neighboring curves differ from each other only by a 0.1% change in the Bubble Radius.

Observing Fig. 8, note that the shape of the tuning curve is only slightly affected by the change in radius. However, the pressure tuning (i.e., the atmospheric pressure where the compliance peak occurs) strongly depends on the Bubble Radius. Consequently, it is a parameter that must be tightly controlled.

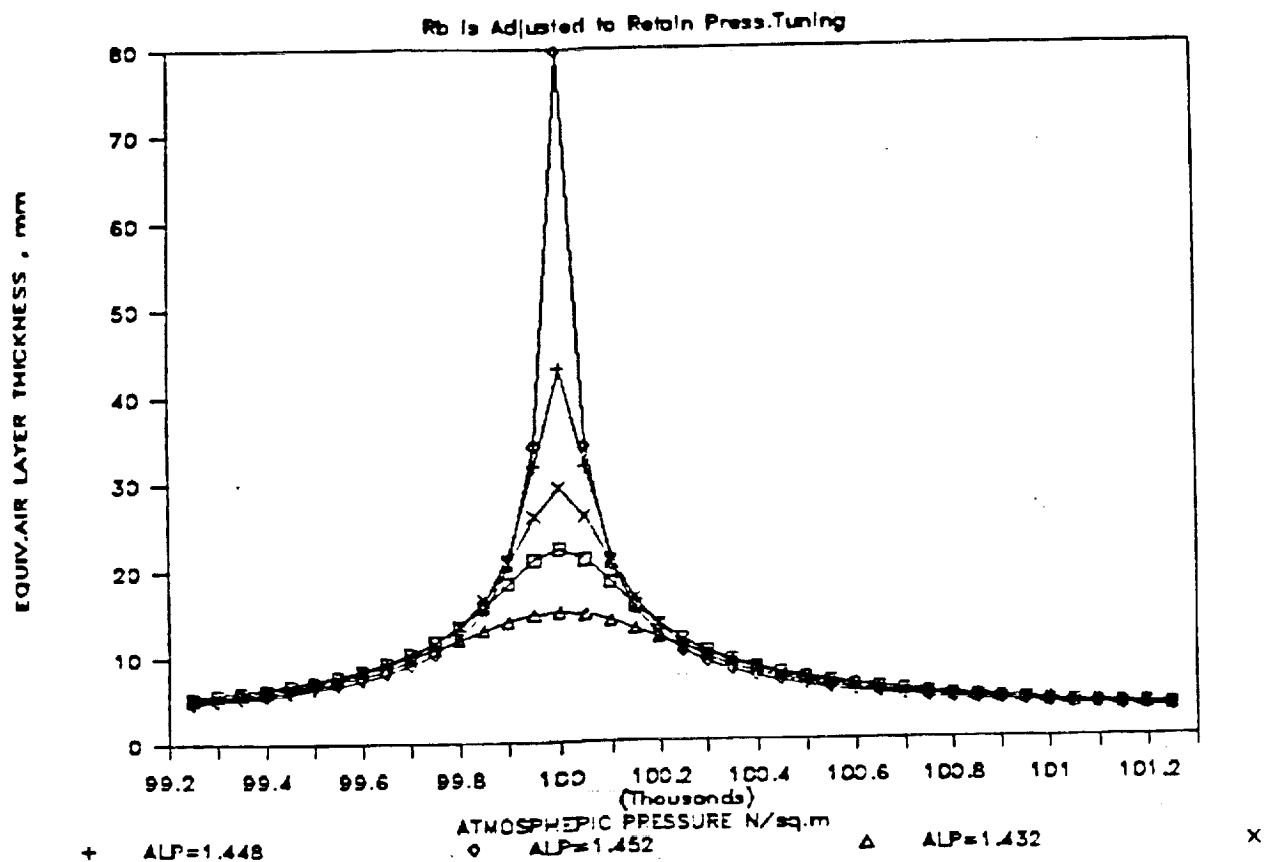


FIGURE 7: EFFECT OF ATMOSPHERIC PRESSURE AND THE $ALP = h/s$ RATIO ON COMPLIANCE.

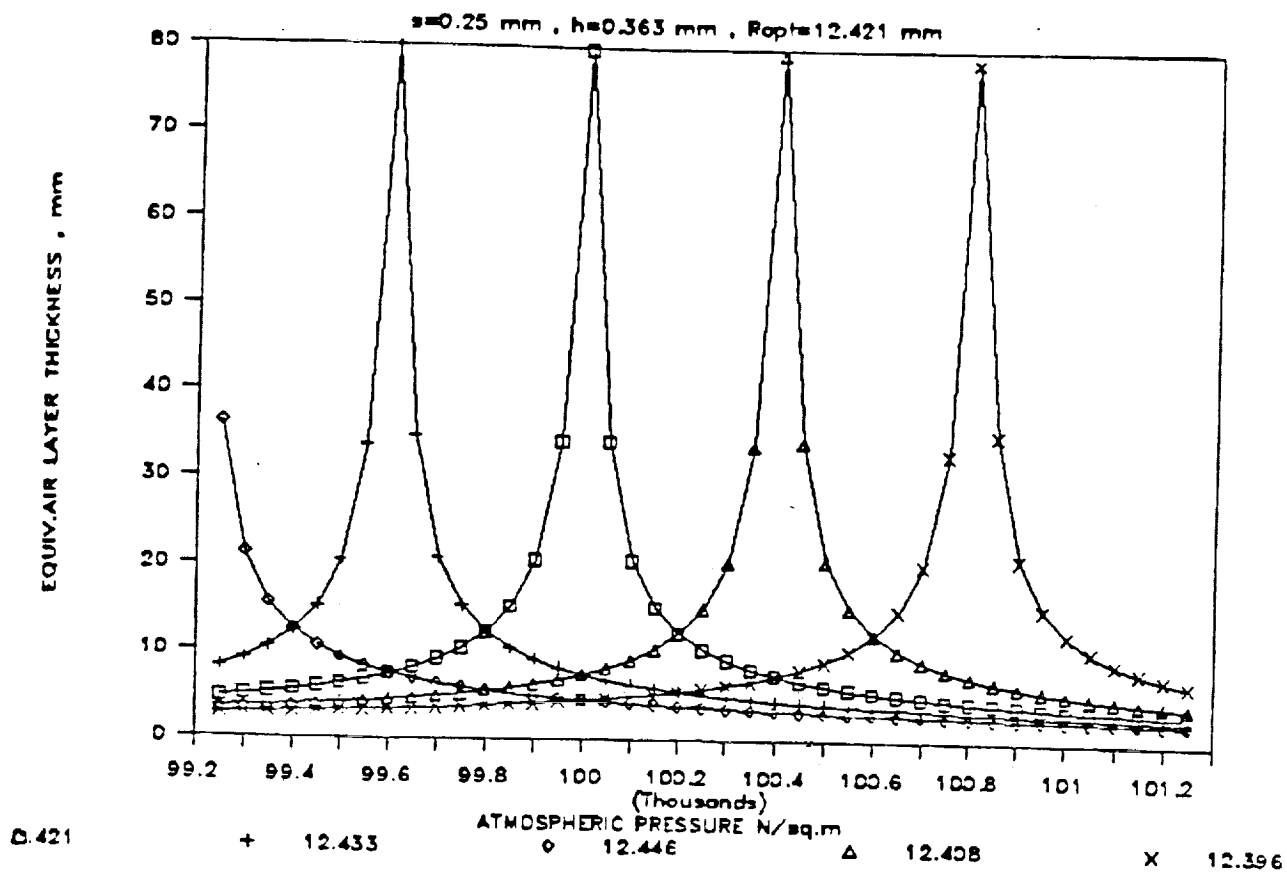


FIGURE 8: EFFECT OF CHANGE IN BUBBLE RADIUS, R_b .

The findings of the parametric studies are summarized in the table below.

TABLE 1
EFFECT OF PARAMETER CHANGES ON PERFORMANCE

Parameter	Larger Than Optimal	Small Than Optimal
Static Pressure ΔP_{tot}	Reduces dynamic compliance, increases resonance frequency	Reduces dynamic compliance, increases resonance frequency
Bubble Radius R_o	Tunes at lower total pressure	Tunes at higher total pressure
Bubble Height h	Tunes at lower total pressure	Tunes at higher total pressure
Bubble Wall Thickness s	Tunes at higher total pressure	Tunes at lower total pressure
h/s Ratio	$h/s > 1.457$, results in unstable conditions (oil canning)	$h/s < 1.457$, tunes at higher total pressure, decreased peak performance

5. KEY ADVANTAGES AND DISADVANTAGES

All traditional sound absorbers and sound attenuating devices have the major drawback that good low frequency performance can be achieved only with a large volume. This is because the reactive part of their impedance at low frequencies is controlled by the stiffness of the air volume, and low stiffness is achievable only if the volume is large.

Vacuum Bubbles have no air in their cavities. Consequently, they do not need a large volume to achieve a low dynamic stiffness. The dynamic stiffness is controlled by the dynamics of the bubble wall alone. To achieve a low dynamic stiffness of the bubble wall, use is made of the nonlinear load/deflection characteristics of a shell that is just "warped" to the proper extent by the static pressure.

The key advantage of the Vacuum Bubble, that very large dynamic compliance is achievable with a very small volume, is accompanied by certain key disadvantages. Both the advantages and disadvantages are discussed below.

5.1 Advantages

Vacuum Bubbles have the following advantages:

- (1) They require substantially less volume to achieve a high dynamic compliance than traditional acoustic elements such as Helmholtz resonators, quarter-wavelength sidebranch resonators, porous sound-absorbing layers, etc.
- (2) Below the resonance frequency of the Vacuum Bubble, the compliance is independent of frequency. As is the case with all resonators, the compliance reaches extremely high values at the resonance frequency. With judicious choice of the

material, geometric and environmental parameters, a Vacuum Bubble can be designed to yield a specified low resonance frequency (and correspondingly a very large dynamic compliance at this frequency) with a very small volume.

- (3) Because Vacuum Bubbles are made of a thin metallic shell and have vacuum in their cavities, they pose no danger to contaminate the environment or pose a fire hazard.
- (4) If used as springs in vibration isolation applications, Vacuum Bubbles require substantially smaller volume, height and static deflection than an airspring that would provide the same dynamic compliance.

5.2 Disadvantages

The key disadvantage of the Vacuum Bubble is that the nearly "miraculous" property of being able to achieve a very high dynamic compliance with a unit of very small volume occurs only under specific conditions. Small changes in these conditions result in a large reduction in the acoustic performance. The disadvantages that all stem from the high sensitivity of performance on environmental, geometric and material parameters are listed below in the perceived order of importance:

- (1) The dynamic compliance, and consequently also the resonance frequency, is very sensitive to the variation of the static pressure.
- (2) Material parameters such as density, Young's modulus, Poisson ratio and geometric parameters, such as wall thickness, bubble height and bubble radius all must be controlled to a very high accuracy to achieve optimal performance.

- (3) As a consequence of the first listed, undesirable property, a single Vacuum Bubble, or a Vacuum Bubble screen or liner made of exactly the same units, fails to provide desired performance if the static pressure varies even if the variation is the normal range of atmospheric pressure variations.

If the static pressure varies in a specific range, a group of differently pressure-tuned Vacuum Bubbles is needed to achieve a specific acoustic performance. Since manufacturing inaccuracies are expected to result in a random variation of the pressure-tuning of nominally identical Vacuum Bubbles, it may be feasible to select specific samples from a single batch of manufactured units which, as a group, would perform well even if the atmospheric pressure varies within a specified narrow range.

6. POTENTIAL APPLICATIONS AND EXPECTED PERFORMANCE

This section contains a brief discussion of the potential use of Vacuum Bubbles in various noise and vibration control applications. Formulas for estimating expected performance are also provided.

6.1 Reduction of Sound Radiation from Large Vibrating Surfaces

If a large vibrating surface is covered by a Vacuum Bubble Sheet, that has a lower acoustic impedance than the characteristic impedance of the surrounding fluid, then it is easier to "squish" the Vacuum Bubble than to compress the fluid. Consequently, the sound power radiated into the fluid is reduced.

The reduction in sound power level, ΔPWL , for an in-phase normal vibration of the radiating surface is given by

$$\Delta\text{PWL} = 10 \log (1 + \rho_0 c_0 / Z'_{ab}) \quad (26)$$

where $\rho_0 c_0$ is the characteristic impedance of the surrounding fluid and $Z'_{ab} = Z_{ab} S_{\text{tot}}$ is the total acoustic impedance of the Vacuum Bubble screen covering the vibrating surface of area S_{tot} . For example, the sound power level reduction for $Z'_a = \rho_0 c_0$ is 6 dB and for $Z'_a = (1/3) \rho_0 c_0$ is 12 dB.

6.2 Reduction of Sound Radiation of Small Sound Sources

If a Vacuum Bubble or Vacuum Bubble group is placed in the immediate vicinity of a constant volume velocity point source, the compliant bubbles deform and "capture" part of the volume flow of the source. Consequently, the sound radiated into the surrounding medium is reduced. If the sound source has a radiation impedance Z_{rad} , and the Vacuum Bubble or Vacuum Bubble

group, the acoustic impedance Z_{ab} , then the resulting radiation impedance, Z_{rad} , is given by

$$Z'_{rad} = \left(\frac{1}{Z_{rad}} + \frac{1}{Z_{ab}} \right)^{-1} \quad (27)$$

The change in the level of the radiated power, due to the presence of the Vacuum Bubble is

$$\Delta PWL = 10 \log \frac{\operatorname{Re}\{Z'_{rad}\}}{\operatorname{Re}\{Z_{rad}\}} \quad (28)$$

If the radiation impedance has the form

$$Z_{rad} = R_{rad} + jX_{rad} \quad (29)$$

and the acoustic impedance of the Vacuum Bubble has stiffness character and has the form

$$Z_{ab} = -jX_{ab} , \quad (30)$$

then, for the case of $X_{ab} \ll X_{rad}$, the change in the level of the radiated power is given by

$$\Delta PWL = 20 \log \left(\frac{X_{ab}}{|Z_{rad}|} \right) \quad (31)$$

For example, if the acoustic impedance of the bubble is 1/10th that of the absolute value of the radiation impedance, then the level of the radiated power decreases by 20 dB.

The same formulas (i.e., Eqs. 28 and 31) can be used to assess the effect of Vacuum Bubbles on the sound radiation from the open end of pipes and small piston-like vibrating surfaces using the appropriate radiation impedance that characterizes the source.

6.3 Sound Absorption of a Vacuum Bubble Layer

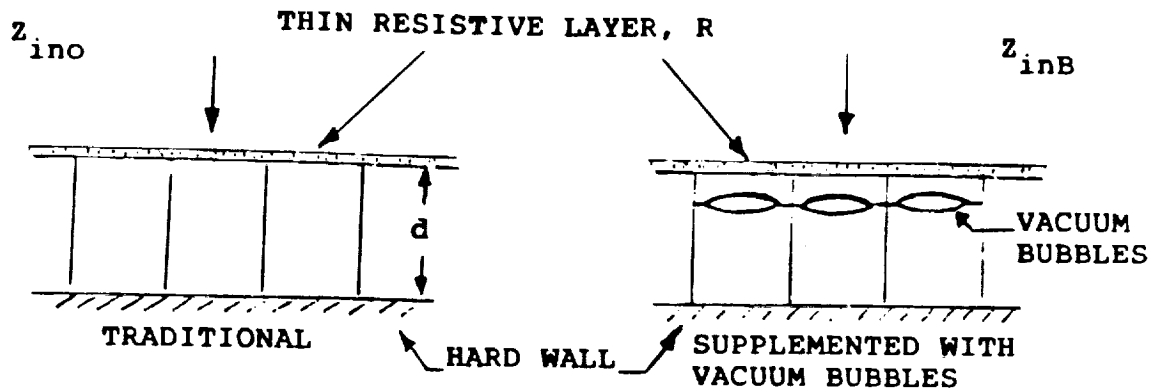
If a plane sound wave is incident on a Vacuum Bubble screen that is placed against a hard, unyielding wall, part of the incident sound power is absorbed. If the Vacuum Bubble screen has a purely resistive acoustic impedance $Z_a = R$, then the sound absorption coefficient, $\alpha(\theta)$ for plane waves incident at an angle θ is given by

$$\alpha(\theta) = \frac{\frac{4R}{\rho_o c_o / \cos \theta}}{1 + \left(\frac{R}{\rho_o c_o / \cos \theta} \right)^2} \quad (32)$$

For example, for $R = \rho_o c_o$ and $\theta = 0$ the absorption coefficient is unity indicating that all of the incident sound energy is absorbed. For $R = \rho_o c_o$ and $\theta = 45^\circ$ the absorption coefficient is $\alpha(45^\circ) = 0.97$. Vacuum Bubbles lend themselves especially well for absorbing low frequency sound because they require substantially less volume than Helmholtz resonators or thick porous layers.

6.4 Supplementing Traditional Sound Absorbers

A frequently-employed traditional sound absorber consists of a thin layer of porous resistive material, such as felt metal, placed at a distance, d , from a hard wall as illustrated schematically in the sketch on the left below.



The disadvantage of the traditional sound absorber is that at low frequencies the stiffness of the airtight layer between hard wall and the resistive layer controls the acoustic wall impedance and the sound absorption coefficient is low, unless the layer thickness is approaching a quarter wavelength. If Vacuum Bubbles are placed in the cavity, as depicted in the right-hand side of the sketch above, they reduce the stiffness and consequently increase the sound absorption at low frequencies. Dividers are needed to prevent sound propagation in the airspace parallel to the surface and thereby rendering the absorber locally reacting.

The normal incidence sound absorption coefficient of the absorber is defined as

$$\alpha_n = 1 - \left| \frac{1 - \rho_o c_o / Z_{in}}{1 + \rho_o c_o / Z_{in}} \right|^2 \quad (33)$$

The input acoustic impedance without and with the Vacuum Bubbles, Z_{ino} and Z_{inB} is given in Eqs. 34a and 34b, respectively.

$$Z_{ino} = R + X_d, \frac{Nsec}{m^3} \quad (34a)$$

$$Z_{inB} = R + \frac{Z_B}{1 + Z_B/X_d}, \frac{Nsec}{m^3} \quad (34b)$$

where

R = specific flow resistance of the resistive layer, $\frac{Nsec}{m^3}$,

Z'_B = acoustic impedance of the vacuum bubble liner behind the resistive layer, $\frac{Nsec}{m^3}$,

d = airspace thickness, m,

$X_d = j\rho_0 c_0 \cot(k_0 d), \frac{Nsec}{m^3}$, and

$k_0 = \omega/c_0$ = is the wavenumber, $\frac{1}{m}$.

At low frequencies, where $k_0 d \ll 1$

$$X_d = -j \frac{P_0 \kappa}{\omega d}, \frac{Nsec}{m^3} \quad (35a)$$

$$Z_B = -j \frac{P_0 \kappa}{\omega d_{eq}}, \frac{Nsec}{m^3}$$

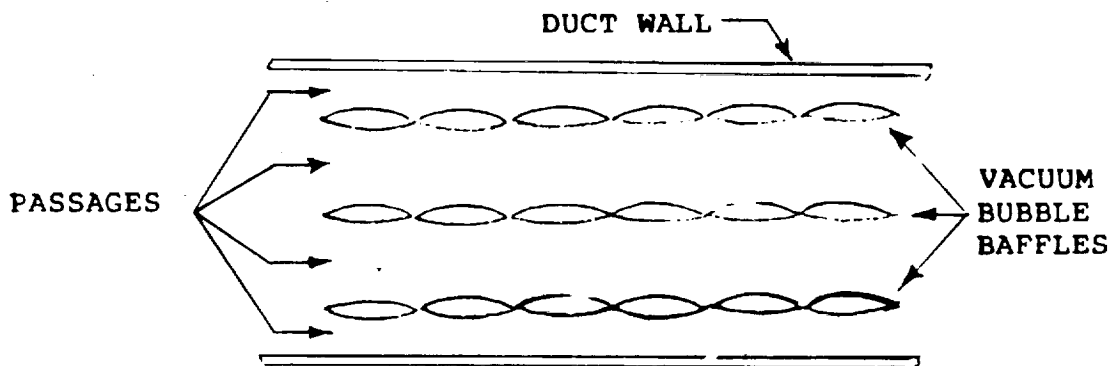
and d_{eq} is the equivalent air layer thickness of the vacuum bubble sheet as defined in Eq. 10. Inspecting Eq. 34b, one notes that, in respect of increasing compliance, placing the vacuum bubble in the airspace is equivalent to an increase of the airspace thickness from d to $d + d_{eq}$.

6.5 Use in Silencers Baffles

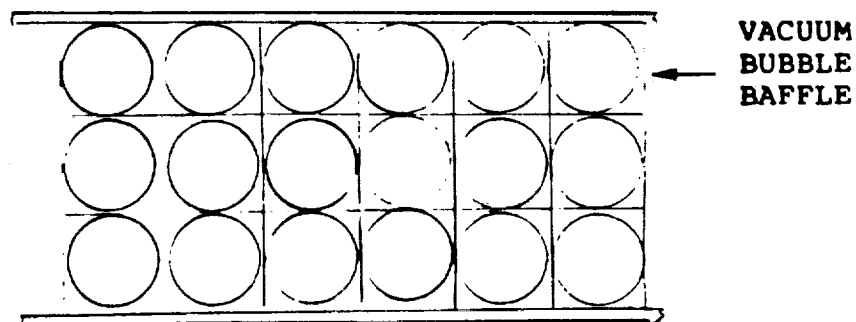
There are two potential uses of Vacuum Bubbles in Silencer Baffles. Thin Vacuum Bubble Sheets may be used as sound-absorbing baffles or Vacuum Bubbles may be used to decrease the air stiffness in a conventional silencer baffle to increase performance at low frequencies. Both of these potential applications are discussed below.

Vacuum Bubble Sheet as Parallel Baffles

As depicted in the conceptual sketch below, Vacuum Bubble Sheets may form baffles of a silencer.



TOP VIEW OF PASSAGE



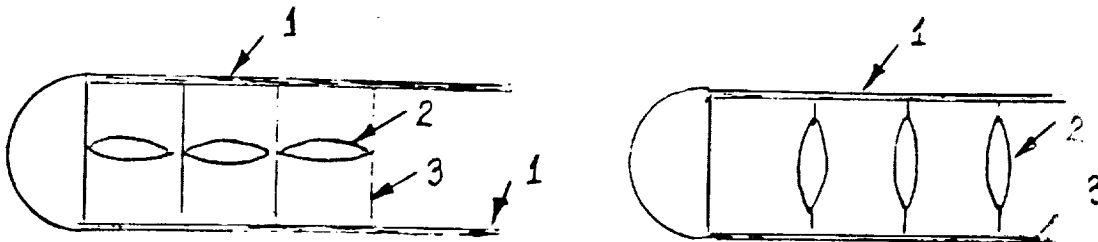
SIDE VIEW OF PASSAGE

In this case, the wall impedance is the acoustic impedance of the Vacuum Bubble Sheet. Because the Vacuum Bubble sheets are locally reacting, their acoustic behavior is fully characterized by the acoustic impedance. Sound attenuation performance can be computed by using the theory developed for predicting the sound attenuation of parallel baffle silencers with locally reacting wall impedance. The improved low frequency performance is due to the reduction of the air stiffness. The Vacuum Bubble baffles are preferably so designed that the resonance frequency of the bubbles falls in the frequency range where most attenuation is needed.

A major obstacle for employing Vacuum Bubble sheets as parallel baffles in silencers is that they are directly exposed to turbulent flow and static pressure differences due to variation of the local flow velocity. Changes in static pressure "detunes" the bubbles, shifting them into a static pressure range that may fall outside of the design static pressure where the bubbles are most compliant.

Vacuum Bubbles Supplementing Traditional Silencer Baffles

As already discussed in Sec. 6.4, Vacuum bubbles can be used to decrease the imaginary part of the wall impedance by reducing the stiffness of the air volume behind the porous resistive surface layer in sound absorbers or silencer baffles. The Vacuum Bubbles may be placed in the muffler baffle in a similar manner than that depicted in the sketch in Sec. 6.4. Another, structurally and functionally more convenient, arrangement of the Vacuum Bubbles is depicted in the conceptual sketch below.



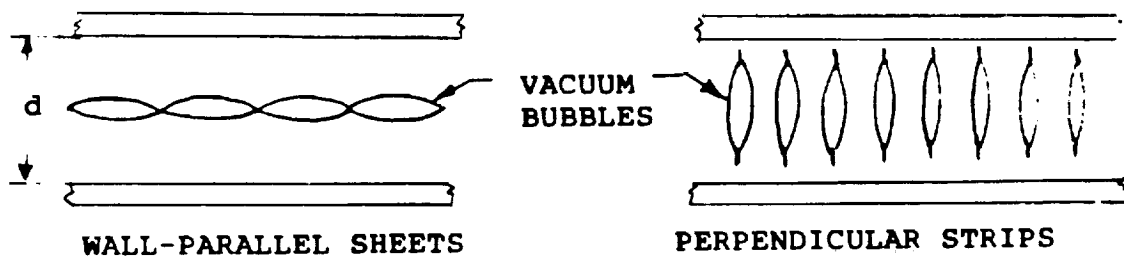
1-Thin Resistive Layer; 2-Vacuum Bubble; 3-Spacer

This arrangement has the following advantages:

- (1) The Vacuum Bubble Strips serve as dividers rendering the baffle locally reacting;
- (2) A much larger number of Vacuum bubbles can be accommodated per unit baffle surface than it would be possible for wall-parallel orientation. This is especially important if the baffle must work in a wider static pressure range requiring groups of differently pressure-tuned bubbles.
- (3) The Vacuum Bubbles are protected from direct flow impingement by the porous, resistive surface layer.

6.6 Increasing the Sound Transmission Loss of Double Walls at Low Frequencies

The sound transmission loss of double walls at low frequencies is controlled by the stiffness of air layer between the two wall panels. Placing Vacuum Bubbles in the cavity results in a substantial decrease of the air stiffness. The bubbles do not absorb or reflect sound, but represent a substantially higher compliance than that of the air volume they replace. As a consequence, the same acoustic performance can be achieved at low frequencies with a double wall of substantially less total thickness or total weight. The conceptual sketches below shows two different ways to insert Vacuum Bubbles into the cavity of a double wall.



The increase in sound transmission loss at low frequencies can be approximated as

$$\Delta TL = 20 \log \left(1 + \frac{d_{eq}}{d} \right)$$

(36)

where

d = wall spacing, and

d_{eq} = equivalent air layer thickness of the vacuum bubbles in the cavity.

Note that d_{eq} is very sensitive to static pressure. In situations where the static pressure experienced by the Vacuum Bubbles undergoes changes, the performance would not be optimal at static pressures that deviate substantially from the design static pressure of the Vacuum Bubbles. The change in static pressure experienced by the bubbles can be reduced if the cavity of the double wall is air-tightly sealed.

6.7 Use as a Spring

A Vacuum Bubble can also be used as a spring. In this case, the combined effect of the static load and the atmospheric pressure both act on the bubble. Consequently, the bubble must be designed such that under this combined load the sagging of the center of the bubble, z , equals the bubble height, h . This assures a high dynamic compliance for the total design static load. The nonlinear static load vs. static displacement curve of the Vacuum Bubble makes it possible to obtain a very high dynamic compliance while the static compliance is small. In other words, a very large dynamic compliance is obtainable without a large static deflection. Note that Vacuum Bubble springs exhibit this high dynamic compliance only for forces that act normal to the center of the bubble membrane. For dynamic forces acting parallel to the plane of the bubble membrane, the dynamic stiffness is high.

7. Possibility to Eliminate Detuning Due to Changes of Atmospheric Pressure

An interesting possibility to eliminate the sensitivity of changing static pressure on the dynamic compliance of Vacuum Bubbles is to design them for peak performance at a small positive pressure in the cavity that corresponds to difference between the highest and lowest range of the outside static pressure. Connecting the cavity to a pump, that is sensor-actuated to maintain a constant difference between the outside and inside pressure, could assure that changes in outside static pressure would not detune the Vacuum Bubbles. This arrangement could be implemented for a single bubble used as a spring or for entire Vacuum Bubble Sheets which are used in airborne noise control applications.

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Report Documentation Page

1. Report No. NASA CR-181829		2. Government Accession No.		3. Recipient's Catalog No.	
4. Title and Subtitle Potential Uses of Vacuum Bubbles in Noise and Vibration Control			5. Report Date June 1989		
			6. Performing Organization Code		
7. Author(s) Istvan L. Vér			8. Performing Organization Report No. BBN Report 6938		
			10. Work Unit No. 535-03-11-03		
9. Performing Organization Name and Address BBN Systems and Technologies Corporation 10 Moulton Street Cambridge, MA 02138			11. Contract or Grant No. NAS1-16521		
			13. Type of Report and Period Covered Contractor Report		
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Langley Research Center Hampton, VA 23665-5225			14. Sponsoring Agency Code		
15. Supplementary Notes Langley Technical Monitor: Kevin P. Shepherd Final Report					
16. Abstract Vacuum Bubbles are new acoustic elements which are dynamically more compliant than the gas volume they replace, but statically they are robust. They are made of a thin metallic shell with vacuum in their cavity. Consequently, they pose no danger in terms of contamination or fire hazard. The potential of the Vacuum Bubble concept for noise and vibration control has been assessed with special emphases on spacecraft and aircraft applications. The following potential uses were identified: (1) as a cladding, to reduce sound radiation of vibrating surfaces and the sound excitation of structures, (2) as a screen, to reflect or absorb an incident sound wave, and (3) as a liner, to increase low frequency sound transmission loss of double walls and to increase the low frequency sound attenuation of muffler baffles. It was found that geometric and material parameters must be controlled to a very high accuracy to obtain optimal performance and that performance is highly sensitive to variations in static pressure. Consequently, it has been concluded that Vacuum Bubbles have more potential in spacecraft applications where static pressure is controlled more than in aircraft applications where large fluctuations in static pressure are common.					
17. Key Words (Suggested by Author(s)) Acoustics Noise Control Vibration			18. Distribution Statement Unclassified - Unlimited Subject Category 71		
19. Security Classif. (of this report) Unclassified		20. Security Classif. (of this page) Unclassified		21. No. of pages 51	22. Price A04

